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Integration of Space Heating and Hot Water Supply in Low Temperature District Heating

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Abstract

District heating may supply many consumers efficiently, but the heat loss from the pipes to the ground is a challenge. The heat loss may be lowered by decreasing the network temperatures for which reason low temperature networks are proposed for future district heating. The heating demand of the consumers involves both domestic hot water and space heating. Space heating may be provided at low temperature in low energy buildings. Domestic hot water, however, needs sufficient temperatures to avoid growth of legionella. If the network temperature is below the demand temperature, supplementary heating is required by the consumer. We study conventional district heating at different temperatures and compare the energy and exergetic efficiency and annual heating cost to solutions that utilize electricity for supplementary heating of domestic hot water in low temperature district heating. This includes direct electric heating and three heat pump solutions applying R134a and R744. The results show that conventional solutions at lowest possible temperature have the highest exergetic efficiency of 28% and lowest annual cost of € 690 for a 159 m^2 house. The best low temperature system is an R134a heat pump with hot water storage on the district heating side, which reaches 25% exergetic efficiency.

Keywords: Low temperature district heating, space heating, domestic hot water, heat pumps, exergy

1. Introduction

District heating provides a means for efficient heating in urban environments. In particular, the integration with combined heat and power (CHP) plants and heat storage makes it possible to produce power and heat with significant flexibility and at high efficiency. The Danish energy system is an example of a system for which extensive expansion of CHP has contributed to make it possible to not increase the energy consumption of society over the last decades [1].

Figure 1 illustrates a generic district heating system which involves combined heat and power production as the main heat supplier. It supplies district heating to a stratified storage tanks, which is connected to the transmission network in order to fulfil the consumer demand. The transmission network operates at high temperatures and has a high annual utilization of capacity, which results in relatively low heat loss. The heat is transferred to the distribution system via substations. The distribution system works with lower temperatures, but due to smaller pipe

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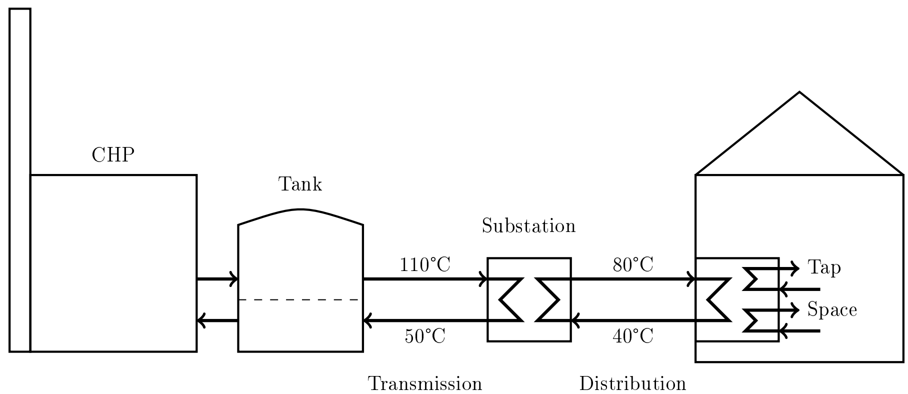

Figure 1. Illustration of district heating system

dimensions and lower capacity, the heat loss is relatively high. Heat is eventually supplied to the consumers by service pipes.

The configuration and performance of the heat supply of a district heating system is governed by a number of criteria. These include:

Energy source The energy input to the district heating system may be fossil fuels, renewable fuels or alternative renewable energy sources. These may be heat sources, e.g., solar or geothermal, or electric power from wind, hydro, or photovoltaics.

For larger systems more heat producers may be connected to the system, such that the heat is based on a mixture of sources.

Conversion Efficiency The conversion efficiency of the energy input to heat supply involves efficiency of the plants, which may be combined heat and power, CHP, or separate heat production.

Heat pumps will most likely play a larger role in the future. For these the Coefficient of Performance, COP, will have an impact on the efficiency of the heat supply.

Heat loss from the network is another important factor. This factor is closely related to the temperature of the forward and return lines of the system.

System Configuration The system may include a number of subsystems which are connected by pipes with a given capacity.

For larger systems with longer distances and with different supply systems on the consumer side, the network will be constituted of a transmission system and distribution systems for each consumer subsystem.

District heating has the potential of being an efficient and cost-effective heat production. A drawback of district heating is, however, the heat loss from transmission and distribution pipes. This is, however, the only energy loss that occurs in the system, and avoiding any heat loss would result in a 100% efficient supply based on first law approach.

But heat loss is not the only source of thermodynamic irreversibilities in district heating. Heat transfer at finite temperature differences and pressure loss in fluid flow results in exergy destruction.

On average the heat loss amounts to about 20 % of the energy consumption for Danish district heating systems [1]. For further developing district heating solutions low temperature transmission

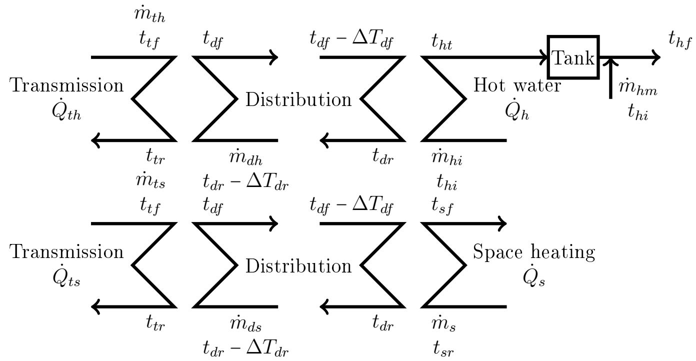

Figure 2. Conventional system

and distribution systems are currently investigated by different researchers as this will increase the efficiency of the total system.

Song [2] develops an exergy-based methodology for investigating the total performance of heating systems based on CHP and heat pumps. It is shown that exergetic efficiency of conventional heating is less than 10 % and that up to 50 % efficiency may be reached by improved systems including waste heat recovery.

Olsen et al. [3] discuss the potential of using low temperature district heating for low energy buildings and the demand for minimum heat loss. In [4] a heating system for space heating with low temperature district heating is investigated. [5] also discusses the potential benefits of low temperature district heating, specifically reduced heat loss, integration of additional heat sources and increased efficiency. Brand [6] describes solutions for low temperature systems for buildings of various standards.

By integrating heat pumps in the district heating system, low temperature sources, e.g., waste heat, may be integrated. The heat pumps may also provide means for lowering the temperatures in the network by boosting the temperatures by the consumers. Lorentzen [7, 8] suggests R744 (CO_2) as an anternative natural refrigerant for low temperature district heating. Several heat pump configurations are investigated by [9].

Low temperature district heating has been investigated based on energy and exergy criteria by e.g., [10]. In [11] the costs of low temperature district heating in terms of both energy and exergy are studied. A detailed study of exergy losses in a complete district heating network is studied in [12]. Pirouti [13] presents a comprehensive study of both high and low temperature district heating.

1.1. District Heating Configurations

District heating may be connected to the consumer system in different ways as illustrated below. The heat demand of the urban environment is assumed to consist of space heating and domestic hot water. The difference between the systems is in the domestic hot water supply, as the space heating system is the same in all cases for a low energy building with floor heating. In the analysis and illustrations, the configurations for both domestic hot water and space heating are considered.

Conventional configuration. A conventional district heating system is illustrated in figure 2. It connects the heating plant(s) and the consumers by a transmission network from the plants to substations and distribution networks from the substations to the consumers. The consumers need heat both for space heating and hot water consumption. The figure illustrates the two supply

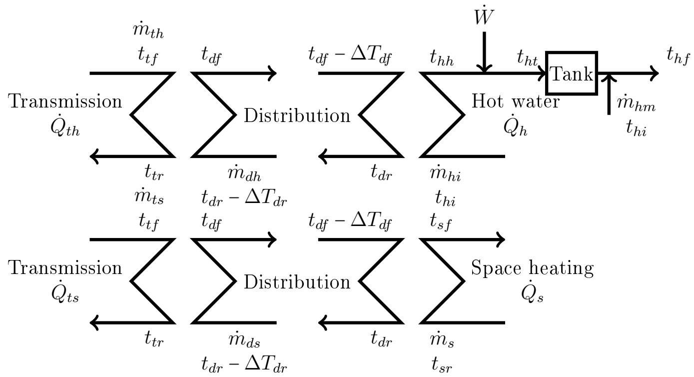

Figure 3. Low temperature system with electric heating of hot water

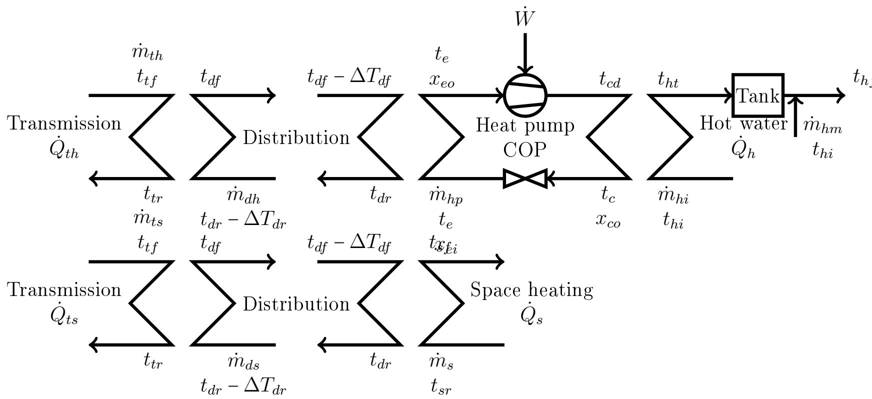

Figure 4. Low temperature system with heat pump and secondary side tank

systems individually, but this is obviously not how the installation would be in practice, as the house is only connected to one supply system from the district heating.

Low temperature configuration with electric heating. Space heating in radiator systems requires temperatures of 50-70 °C, but modern low energy buildings equipped with floor heating makes it possible to lower this temperature to about 30-40 °C. Thus, lower temperature may be possible in the distribution network and accordingly lower heat loss would be acquired. However, if domestic hot water is stored in a tank, it must be heated to 50-60 °C to avoid health risks caused by legionalla bacteria growth. This means that even if the space heating demand can be covered by lower temperature networks, the hot water temperature cannot be avoided. An example of an installation is ilustrated in figure 3. The required hot water temperature is obtained by supplementary electric heating between the district heating and the hot water tank. However, direct electric heating is thermodynamically inefficient. For this reason heat pumps for boosting the hot water temperature are considered. Three heat pump configurations are considered depending on the location of the hot water tank.

Low temperature configuration with heat pump and secondary side tank. The tank may be installed on the consumer side, the secondary side, as illustrated in figure 4. In this configuration the district heating supplies the heat pump evaporator. The heat pump directly heats the domestic hot water, which is stored in a hot water tank. As the fresh water enters at a temperature which is significantly lower than the condenser temperature a significant exergy loss occurs. This makes it relevant to investigate the performance of a transcritical R744 heat pump, which utilizes the temperature glide

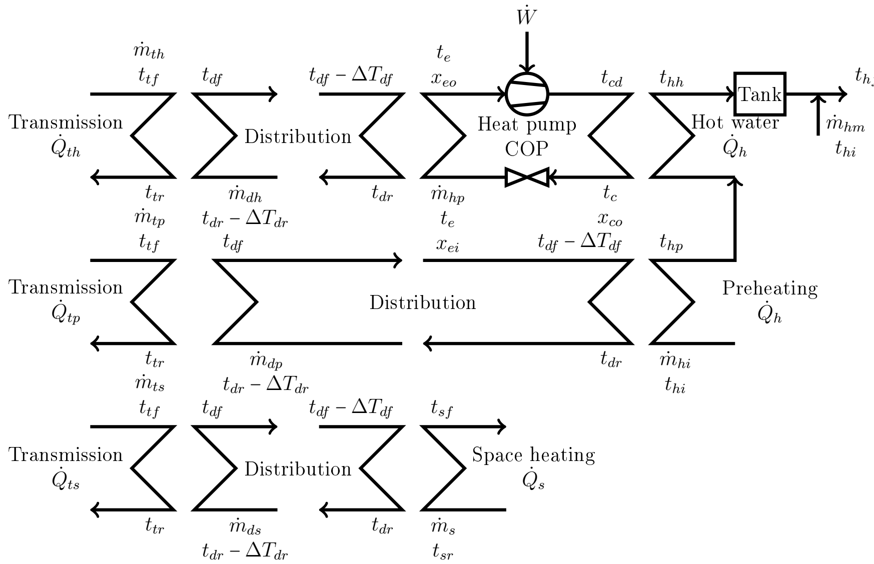

Figure 5. Low temperature system with heat pump, secondary side tank and preheating of domestic hot water

of the working fluid in the gas cooler.

Low temperature configuration with heat pump, secondary side and and preheating. Another way to decrease the irreversibility of the heat transfer is illustrated in figure 5. In this configuration, the fresh water is preheated by district heating before entering the condenser of the heat pump. This configuration would in practice require two storage tanks in order to limit the load on the district heating supply at high demands for hot water.

Low temperature configuration with heat pump and primary side tank. The secondary side hot water tank implies a risk of legionella bacteria growth of the water due to the storage at increased temperature. For this reason, it is required to reach a sufficiently high temperature to eliminate the health risk. This situation may be avoided by a configuration with hot water storage on the district heating side, the primary side, instead. By storing hot water on this side, the fresh water entering the system may be heated by the water in the tank, and no need for storage of the consumer water is involved. This configuration is illustrated in figure 6. As a lower temperature is required in the storage, the heat pump will have a higher efficiency. The district heating supply is both used as the heat source for the heat pump evaporator in the upper part of the configuration, and as the heat sink for the condenser in the middle part of the system and connected to the heat pump by nodes A and B. The temperature in the tank, t_{hh}, is in this case lower than required in the previous configurations and no mixing is made to reach the actual consumer temperature.

1.2. Scope of work

The present paper compares the described concepts for space heating and domestic hot water in conventional district heating and low temperature district heating networks based on annual time-averaged calculations. The development and installation of a heat pump for this purpose is explained in detail in [14], [15] and [16].

Case study for low energy building. The basis of the work is a new settlement of 116 buildings. We study a house of 159 m^2 with 4 inhabitants. It has an annual consumption of 4010 kWh for space heating and 3200 kWh for hot water [14]. The space heating demand occurs about 4000

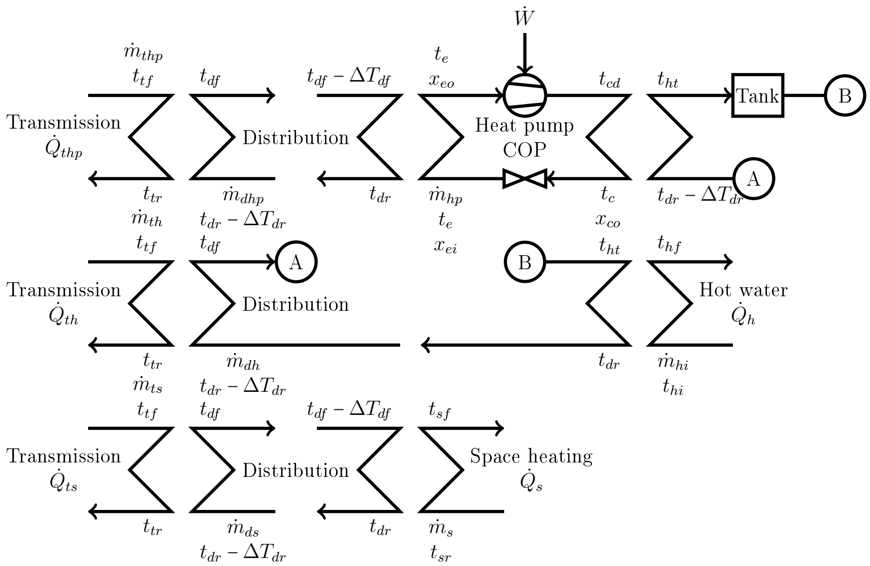

Figure 6. Low temperature system with heat pump and primary side tank

Figure 7. Micro Booster Heat Pump for Hot Water Production

hours annually, whereas hot water is used daily for e.g., showers, dish washing and hand washing. This results in that annual averages of 365 W hot water and 458 W space heating are consumed. Space heating is provided at 30/22°C and hot water at 50/10°C in the house. Hot water is supplied from a tank which may be located on the primary (district heating distribution) side or on the secondary (consumption) side in the system. In the cases where the hot water storage is located on the secondary side a temperature of 60 °C is required in the tank.

Heat loss calculations are based on data and are given as a coefficient per unit pipe length of 65 W/km K in total for both the forward and return sides of the twin pipe. The length of forward and return pipes in the network is 3.6 km. The ground temperature is assumed to be 10 °C. Heat loss is only considered in the distribution network.

The objective of the work is to understand the performance of low-temperature district heating systems compared to conventional systems. The low temperature systems are characterized by the need of supplementary electric heating, e.g., heat pumps, for providing hot water. The work considers the integration of low temperature distribution systems in combination with, possibly existing, transmission systems. This means that the transmission system temperatures are maintained, because it is difficult to change the operation temperatures of such a system. The benefit of low temperature systems is that significantly lower heat loss is obtained while it on the other hands introduces demands for electricity for heat pump operation. The system is assumed to mostly utilize large extraction steam cycle CHP plants as described in e.g., [17] as these are flexible and efficient. If a new system was considered it would be possible to dimension it for low temperature operation. This would mean that the CHP plant would be able to operate at a higher electric efficiency. The effect is however minor as explained in [18].

2. Methods

Based on a screening of the possible configurations, the following cases are studied:

- Conventional system 80/40°C
- Conventional system 65/55°C
- Conventional system 60/30°C
- Low temperature system with electric heating 45/25°C
- Low temperature system with heat pump and secondary side tank 45/25°C
- Low temperature system with heat pump and secondary side tank and preheating 45/25°C
- Low temperature system with heat pump and primary side tank 45/25°C

The heat pump systems are calculated for both a conventional R134a heat pump cycle and a transcritical CO_2 cycle.

The transmission system operates at 85/60°C to cover all cases.

The consumer cost of the energy supply is calculated by estimated electricity and district heating prices of 0.30 €/kWh (2.26 DKK/kWh) [19] and 0.10 €/kWh (0.77 DKK/kWh) [20], respectively, in a conventional system. The benefits resulting of decreased heat loss is assigned to the consumer prices in low temperature scenarios.

2.1. Modeling

The different configurations have been implemented as models in DNA [21]. The models consist of component models which include heat exchangers, distribution pipes and the components of the heat pumps, i.e., evaporator, condenser, compressor and valves. The fundamental laws of thermodynamics are respected in all component models.

The following parameters are used throughout the calculations:

Minimum temperature difference in heat exchangers in cycle	2.5	K
Minimum temperature difference in heat exchangers in network	5	K
Minimum temperature difference in tank coil	5	K
Isentropic efficiency of heat pump compressor	50	%
Heat transfer coefficient from network (distribution and service lines) to ground	1.9	W/K
Ground temperature	10	°C
Pressure loss	0	Pa

for the heat pump compressor heat loss, as well as condenser subcooling and evaporator superheating are neglected.

Heat loss from the network is calculated individually for the forward and return pipes based on the heat transfer coefficient and the temperature difference between water and ground.

2.2. Evaluation of Performance

The results of the calculations are the required heat supply from the transmission system, the electricity demand, the total heat loss and annual cost of heat. The analysis is based on annual averages of heat demands for both space heating and hot water consumption.

The results are quantified in terms of energy, exergy and consumer costs. For energy a unit energy from district heat and electricity are equal, which makes it difficult to define a meaningful efficiency. Contrary, exergy not only measures the energy content, but also takes the "quality" of the energy into account. In thermodynamics exergy is defined as the maximum work that may be extracted from a given amount of energy. This theoretical measure may, however, be described in several ways that show the value of exergy as a measure of quantity and quality of energy. Exergy may namely also be stated to be the part of an amount of energy that can be converted into any other energy form by a thermodynamically reversible process. For, e.g., electric, mechanical, kinetic and potential energy this fraction is unity, whereas for heat and substances at finite temperature the fraction is smaller.

The exergy analysis in this study is based on the approach described in [22].

Exergy will due to irreversible energy conversion be destroyed and is closely related to entropy that is generated, due to the Guoy-Stodola theorem. It states that exergy destruction and entropy generation are proportional:

$$\dot{E}_{\texttt{dest}} = T_0 \dot{S}_{\texttt{gen}} \tag{1}$$

The two quantities are related by the reference temperature. This shows that exergy resembles a state variable when the reference conditions p_0, T_0 are decided. In the present case they are set to 1 bar and 10°C, respectively.

Primary energy sources as fuel, solar, and wind can all be found to be exergy within 5% accuracy. This shows that exergy destruction is also a quantification of the primary energy supply to a system.

Primary energy utilization in the system may thus be evaluated by the exergetic efficiency:

$$\eta_x = \frac{\dot{E}_{\text{prod}}}{\dot{E}_{\text{cons}}} \tag{2}$$

In the present case the exergy consumption is electricity and district heat, the exergetic product is the space heating and domestic hot water.

$$\eta_x = \frac{\dot{E}_{\mathrm{dhw}} + \dot{E}_{\mathrm{sh}}}{\dot{E}_{\mathrm{dh}} + \dot{E}_{\mathrm{el}}} \tag{3}$$

As changes in kinetic and potential energy are neglected and chemical reactions do not occur, only the physical exergy of the flows is calculated. It is given as:

$$\dot{E} = \dot{m}(h - h_0 - T_0(s - s_0)), \text{ where } h_0 \text{ and } s_0 \text{ are found at } (p_0, T_0) \tag{4}$$

In the present case the exergy input to the heat supply system is defined by the transmission network. We thus neglect the exergy destruction in the CHP plant. This means that it is assumed that the heat in the transmission system is assumed to have been produced by a reversible heat engine driving a reversible heat pump that produces the heat in the system at 85°C/60°C. Similarly, the consumed electricity is assumed to be produced by a reversible heat engine. This is naturally not the case, so in order to quantify the primary energy utilization the calculated exergetic efficiencies should be multiplied by the exergetic efficiency of the CHP production to the transmission system. In the Danish energy system, this efficiency is about 45%, presently.

The exergy content of the energy supply to the consumers is 1 kWh exergy/kWh electricity and 0.18 kWh exergy per unit district heat from the transmission grid.

The exergy cost per unit of electricity supply is equal to the electricity price of 0.30 €/kWh. For heat the cost per unit exergy is 0.57 €/kWh based on 0.10 €/kWh energy.

This shows that even though exergy is a common measure of any energy supply, the price of an exergy unit is not the same in practice. In this case heat almost has the double cost compared to electricity based on the exergy content.

For the economic evaluation the cost of district heating per unit energy is kept constant at the production side. This means that the consumer receives the benefit of lower loss, as the cost is lowered proportionally with the heat loss when the network temperature is lowered.

3. Results

3.1. Conventional District Heating

The base case is a conventional high temperature system where all heat is provided by the district heating without supplementary electricity. Figure 8 illustrates the flows and temperatures in this configuration. The energy flows in this configuration are presented in table 1. It is seen that the energy efficiency of the system is 81% which illustrates the heat loss of the distribution network. The exergetic efficiency is 29 % for the hot water supply and 24 % for space heating, and accordingly 27 % in total. The annual cost of the heat is € 740 (5500 DKK).

For a lower temperature conventional system utilizing temperatures of 65 °C and 35 °C for forward and return, respectively, the results are presented in table 2. The heat loss is lower in this configuration and higher efficiencies are obtained, even if the differences are small.

A configuration with even lower temperatures in the distribution network is presented in table 3. Naturally higher efficiency and lower cost is found is this case. However, the case is not fully applicable, as the requirement of 60 °C cannot be respected in the tank. The case is only acceptable if lower temperature can be allowed, without compromising the health concerns.

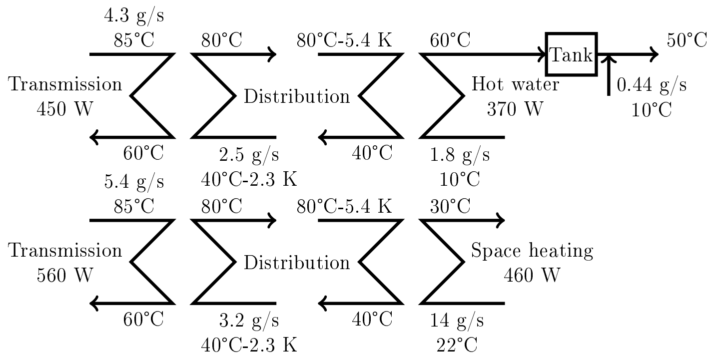

Figure 8. Conventional system at 80/40°C

Table 1. Conventional system 80/40 °C

	Energy [W]	Exergy [W]	Price [€/y]
Hot water			
Hot water supply	370	23	330
Heat loss	81	13	
District heat consumption	450	81	330
Efficiency [%]	*82*	*29*	
Space heating			
Heat supply	460	25	410
Heat loss	110	17	
District heat consumption	570	100	410
Efficiency [%]	*81*	*24*	
Total			
Heat supply	820	48	740
Heat loss	190	30	
Heat consumption	1000	180	740
Power consumption	0	0	
Efficiency [%]	*81*	*27*	

Table 2. Conventional system 65/35 °C

	Energy [W]	Exergy [W]	Price [€/y]
Hot water			
Hot water supply	370	23	320
Heat loss	66	8.8	
District heat consumption	430	78	320
Efficiency [%]	*84*	*30*	
Space heating			
Heat supply	460	25	400
Heat loss	85	11	
District heat consumption	540	98	400
Efficiency [%]	*84*	*25*	
Total			
Heat supply	820	48	710
Heat loss	150	20	
Heat consumption	970	180	710
Power consumption	0	0	
Efficiency [%]	*84*	*27*	

Table 3. Conventional system 60/30 °C

	Energy [W]	Exergy [W]	Price [€/y]
Hot water			
Hot water supply	370	23	310
Heat loss	58	7	
District heat consumption	420	76	310
Efficiency [%]	*86*	*31*	
Space heating			
Heat supply	460	25	390
Heat loss	74	9.2	
District heat consumption	530	96	390
Efficiency [%]	*86*	*26*	
Total			
Heat supply	820	48	690
Heat loss	130	16	
Heat consumption	950	170	690
Power consumption	0	0	
Efficiency [%]	*86*	*28*	

Table 4. Low temperature system 45/25 °C with electric heating

	Energy [W]	Exergy [W]	Price [€/y]
Hot water			
Hot water supply	370	23	590
Heat loss	41	3.7	
District heat consumption	260	47	190
Electricity consumption	150	150	390
Efficiency [%]	*89*	*12*	
Space heating			
Heat supply	460	25	390
Heat loss	74	9.2	
District heat consumption	530	96	390
Efficiency [%]	*89*	*26*	
Total			
Heat supply	820	48	970
Heat loss	180	13	
Heat consumption	790	140	590
Power consumption	150	150	390
Efficiency [%]	*89*	*16*	

3.2. Low Temperature District Heating 45/25°C with Electric Heating

For the simplest low temperature configuration using electric heating the results are presented in table 4. It is found that a significant share of electricity is required to reach the target domestic hot water temperature, i.e., 150 W out of the 370 W demand. This results in low exergetic efficiency of the hot water heating, and an efficiency of only 16 % of the total system. The annual cost is € 970 (7300 DKK).

3.3. Low Temperature District Heating with Heat Pump

The three different heat pump configurations reach higher efficiency than the simple low temperature system with electric heating.

Secondary side tank. With the hot water tank on the consumer side an exergetic efficiency of 20 % is obtained with R134a as shown in table 5. The heat pump has a COP of 4.2 and consumes 86 W power.

For the system with R744, table 6, instead higher efficiency results are obtained. The exergetic efficiency is 24 %. This heat pump reaches a COP of 6.6 and consumes only 56 W power.

The preheat configuration with R134a is competitive with R744 and reaches an efficiency of 24 % as well, as presented in table 7. The heat pump COP is still 3.5 but it is utilitzed for higher temperature heating and thus only consumes 41 W.

The annual cost of heat for the three configurations are € 820 (DKK 6100), € 800 (DKK 5900) and € 760 (DKK 5700), respectively. The lowest electricity consumption is thus most attractive.

The configuration has a minimum temperature difference of 2.5 K between the fluids in the gas cooler.

Primary Side Tank. The best configuration for the low temperature system is the configuration with hot water storage on the district heating side. The calculations are presented in table 8.

Table 5. Low temperature system at 45/25°C with R134a and secondary side tank

	Energy [W]	Exergy [W]	Price [€/y]
Hot water			
Hot water supply	370	23	450
Heat loss	41	3.7	
District heat consumption	280	58	210
Electricity consumption	86	86	230
Efficiency [%]	*90*	*16*	
Space heating			
Heat supply	460	25	390
Heat loss	74	9.2	
District heat consumption	530	96	390
Efficiency [%]	*89*	*26*	
Total			
Heat supply	820	48	820
Heat loss	120	13	
Heat consumption	810	150	590
Power consumption	86	86	230
Efficiency [%]	*92*	*20*	

Table 6. Low temperature system at 45/25°C with R744 and secondary side tank

	Energy [W]	Exergy [W]	Price [€/y]
Hot water			
Hot water supply	370	23	400
Heat loss	41	3.8	
District heat consumption	350	63	260
Electricity consumption	56	56	150
Efficiency [%]	*90*	*20*	
Space heating			
Heat supply	460	25	390
Heat loss	74	9.2	
District heat consumption	530	96	390
Efficiency [%]	*89*	*26*	
Total			
Heat supply	820	48	800
Heat loss	120	13	
Heat consumption	880	140	650
Power consumption	56	56	150
Efficiency [%]	*88*	*24*	

Table 7. Low temperature system at 45/25°C with R134a and secondary side tank and preheating of domestic hot water

	Energy [W]	Exergy [W]	Price [€/y]
Hot water			
Hot water supply	370	23	370
Evaporator side			
Heat loss	17	1.5	
District heat consumption	120	22	90
Preheating			
Heat loss	25	2.3	
District heat consumption	240	44	180
Heat Pump			
Electricity consumption	38	38	100
Efficiency [%]	*90*	*22*	
Space heating			
Heat supply	460	25	390
Heat loss	74	9.2	
District heat consumption	530	96	390
Efficiency [%]	*89*	*26*	
Total			
Heat supply	820	48	750
Heat loss	120	13	
Heat consumption	890	160	650
Power consumption	38	38	100
Efficiency [%]	*89*	*24*	

Table 8. Low temperature system at 45/25°C with R134a and primary side tank

	Energy [W]	Exergy [W]	Price [€/y]
Hot water			
Hot water supply	370	23	350
Evaporator side			
Heat loss	12	1.0	
District heat consumption	130	24	97
Condenser side			
Heat loss	28	2.4	
District heat consumption	240	44	180
Heat pump			
Electricity consumption	29	29	76
Efficiency [%]	*90*	*24*	
Space heating			
Heat supply	460	25	390
Heat loss	74	9	
District heat consumption	530	96	390
Efficiency [%]	*89*	*26*	
Total			
Heat supply	820	48	740
Heat loss	110	12	
Heat consumption	930	170	660
Power consumption	29	29	76
Efficiency [%]	*87*	*25*	

The heat pump has a COP of 9.6 and consumes only 29 W to produce domestic hot water. The exergetic efficiency of the system is 25 % and is thus almost competitive with the conventional system. The annual cost of heat is € 740 (DKK 5600).

3.4. Overall Results

In summary the results of the above calculations are given in table 9.

The results show that the exergetic efficiency of the conventional system configurations is higher than in the low temperature cases. This is caused by the low exergy content of heat at the relatively low temperatures in the system. However, the difference in efficiency is low if compared to the best low temperature solutions which are R134a heat pump with primary side tank and with secondary side tank and preheating. The former is considered to be the best solution and it will reach exergetic efficiency of the same values as the conventional system if the minimum temperature differences in the heat pump evaporator and condenser are lowered to 2.5 K.

It should also be noted that the best solution with a 60/30°C distribution network actually involves a temperature crossover, which means that it is not a realisable solution without allowing a little lower temperature in the storage tank.

The three latter heat pump solutions are close in performance. The R744 system may be competitive even though the efficiency is lower than for the R134a solutions. R744 has several advantages such as being a natural refrigerant with very low global warming potential (GWP), a low safety classification and a low price. Other refrigerants that do not operate in a transcritical cycle may have similar performance as R134a and may thus also be competitive.

Table 9. Summary of results

System	Distribution temperatures [°C]	Refrigerant	Heat pump COP [–]	Tank location	Preheating	Heat consumption [W]	Electricity consumption [W]	Energy Efficiency [%]	Exergetic Efficiency [%]	Cost [€/y]	CO_2 emissions [Mg/y]
Conv. 1	80/40	–	–	Sec		1000	0	81	27	740	1.0
Conv. 2	65/55	–	–	Sec		970	0	84	27	710	1.0
Conv. 3	60/30	–	–	Sec		950	0	86	28	690	1.0
LT EL	45/25	–	1.0	Sec		790	150	89	16	970	1.4
LT HP 1	45/25	R134a	4.2	Sec		830	100	92	20	820	1.3
LT HP 2	45/25	R744	6.6	Sec		880	56	88	24	800	1.1
LT HP 3	45/25	R134a	3.5	Sec	×	890	41	89	24	750	1.1
LT HP 4	45/25	R134a	9.6	Prim	×	930	29	89	25	740	1.1

Regarding climate impact of the heat supply for the building, the conventional systems all have a CO_2 emission of close to 1000 kg per year based on the average emission for power and district heating in the Danish energy statistics [1]. The emissions of the best heat pump solutions are close to this value, while the solutions with low efficiency may reach up to 40% higher emissions.

3.5. Sensitivity Analysis

The most relevant heat pump-based solution is the LT HP 3, with a conventional hot water tank. This has been analyzed in further detail to investigate sensitivity of the results with respect to the assumed values. None of the investigated parameters have a very significant impact on the annual cost or the exergetic efficiency.

Low electricity prices is a quite important factor. If the price was 33% lower it would benefit the heat pump solutions, and the annual cost would be 730 €.

Out of the technical parameters, the estimated value of isentropic efficiency of the compressor is conservative, but not unrealistic, as it is based on the component used in the demonstration unit. If it reaches 80% the annual cost would be reduced to 740 € and the exergetic efficiency 25%.

A similar result would be found if the supply temperature of the hot water was reduced to 45°C and all the condenser temperautre was reduced accordingly.

If the heat loss in the network was reduced 50% the cost would be 750€.

Heat pump parameters as evaporator temperature, condenser temperature, and evaporator superheat all have little significance for the annual result and exergetic efficiency.

4. Discussion

The results show that the low losses of the low temperature systems are not sufficient to make these competitive regarding cost or exergy utilization, when compared to conventional systems, even at high temperatures. This is to some extent caused by the high share of domestic hot water in the low energy building.

The exergy flows illustrate the differences due to the lower temperatures through the system. The exergetic efficiency of a modern combined heat and power plant is on the order of 45%, which means that the overall exergetic efficiency of the district heating system is as low as 10%. This shows that there is a significant room for improvement, if the whole chain of energy conversion from fuel to low temperature consumption by the consumer is considered. This involves several heat transfer processes which result in exergy destruction or loss. If low temperature was introduced in the transmission system as well as in the distribution system, lower exergy destruction would be introduced. However, the exergy difference is likely not sufficient to compensate the difference in efficiency fully.

Another consequence of lowering the temperatures in the transmission systems would be that higher electric efficiency could be obtained from the power plant. A similar improvement would be found in smaller systems with distribution network only.

Improved performance of low temperature district heating may be obtained by integrating it with cheap heat sources which are not possible to use at the temperatures in conventional systems. This idea has been suggested by e.g., [23]. [24] integrates industrial waste heat in district heating by use of absorption heat pumps. Other cheap, low temperature heat sources may also benefit the low temperature system, e.g., waste heat, flue gas condensation from biomass-based power and solar heating.

The risk of legionella has in this study been assumed to be avoided by reaching a temperature of 60 °C in domestic hot water tanks. The actual temperature requirement varies in different studies. However, the results are considered to be similar if lower temperatures is required, even if the systems with storage on the consumer side will be improved. Other means of legionella treatment are being developed. They may make it possible to avoid the high temperature of the hot water, but they will require cost for investment and operation.

The prices used in calculations are based on average numbers in Danish conditions. In other countries or in future situations, other values may occur such that electric may be favoured compared to district heating. On the other hand, future systems with improved building standards will require less space heating, and thus the domestic hot water will have an increased share of the demand. From an efficiency viewpoint it would be better to consider heating at different temperatures, such as floor heating and domestic hot water, as two different products, which might be produced in separate systems.

The calculations are based on yearly average consumption rates. This is an important assumption, but it is assumed that taking the seasonal variations into account would not change the overall picture significantly. However, in practical heating installations, the high temperature needed for legionella treatment may only be obtained e.g. weekly. This solution is beneficial for the heat pump solutions as the COP, and thus the power consumption, is directly linked to the water temperature.

The calculations are based on large district heating networks for cities including both transmission and distribution networks with heat production from primarily extraction steam cycle CHP units, but also back-pressure mode cycles. Such systems account for the largest share of the district heating in Denmark. These will have an increase in efficiency if lower temperatures are used in the district heating system, in particular the forward lines. This benefit is not taken into account, but it will not impact the results significantly, as only minor efficiency improvement will be obtained [18]. The transmission system will usually be difficult to change. For other CHP plant types and for smaller networks with single CHP units, other options for improvement may be utilized by lower temperatures in the district heating system. Such systems may benefit more from low temperature solutions and heat pump integration.

Table 9 presents the COP of the heat pump cycle of each of the systems. These are not the *System COP* of the domestic hot water heat pump system, which might be introduced as well. For the system with primary side tank this would be as high as 12, as 370 W heat are supplied by consumption of 30 W electricity. However, the value should be taken with a grain of salt, as COP is always dependent on the exact system configuration. For instance, the electric heating system apparently has a System COP of 2.5 which is not intuitive.

5. Conclusion

Eight configurations for supply of heat for tap water and space heating in a district heating system have been studied to determine the demands for heat and electricity supply. It is found that the systems all have a primary energy utilization, i.e., exergetic efficiency, of 16-28%, and thus that significant potential for improvement theoretically exists. The exergy utilization in the presented systems does not include the CHP production, so the primary energy utilization is only about half of the presented values. The cost of heat is annually between € 690 and € 970. The cost is not proportional to the exergetic efficiency, but the trends are similar.

Conventional systems with higher temperatures in the network have a better utilization than low temperature solutions, as the decrease in heat loss does not compensate the electricity demand to cover the energy consumption. However, district heating will probably in general be converted to lower temperature and thus a heat pump solution is required.

In the complete district heating system, the largest sources of exergy destruction are due to the fuel conversion at the CHP plant and the heat transfer in substations and by consumers. The exergy loss due to heat loss is only a minor contributor to the overall irreversibility.

The results show that a solution with R134a, or other subcritical systems, with heat storage on the primary side, will have the lowest primary energy consumption. A transcritical R744 solution or an R134a solution with preheating both with heat storage on the primary side may also be considered as they have similar performance.

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Nomenclature

COP Heat pump Coefficient of Performance [–]

$\dot{E}$ Exergy flow [W]

$\dot{E}_{\mathtt{dest}}$ Exergy destruction rate [W]

$\dot{E}_{\text{cons}}$ Exergy consumption flow [W]

$\dot{E}_{\text{dhw}}$ Exergy flow of domestic hot water [W]

$\dot{E}_{\text{dh}}$ Exergy flow of district heat supply [W]

$\dot{E}_{\text{el}}$ Exergy flow of electricity [W]

$\dot{E}_{\mathrm{prod}}$	Exergy product flow [W]
$\dot{E}_{\mathrm{sh}}$	Exergy flow of space heating [W]
η_x	Exergetic efficiency [–]
h	Specific enthalpy [kJ/kg]
h_0	Dead state specific enthalpy [kJ/kg]
$\dot{m}$	Mass flow [kg/s]
$\dot{m}_{dhp}$	Mass flow for heat pump heating in distribution [kg/s]
$\dot{m}_{dh}$	Mass flow for hot water in distribution [g/s]
$\dot{m}_{dp}$	Mass flow for preheating in distribution [g/s]
$\dot{m}_{ds}$	Mass flow for space heating in distribution [g/s]
$\dot{m}_{hi}$	Mass flow of fresh water to hot water tank [g/s]
$\dot{m}_{hm}$	Mass flow of fresh water for mixing to supply temperature [g/s]
$\dot{m}_{hp}$	Heat pump refrigerant mass flow [kg/s]
$\dot{m}_s$	Mass flow for space heating in house [g/s]
$\dot{m}_{thp}$	Mass flow for heat pump heating in transmission [kg/s]
$\dot{m}_{th}$	Mass flow for hot water in transmission [g/s]
$\dot{m}_{tp}$	Mass flow for preheating in transmission [g/s]
$\dot{m}_{ts}$	Mass flow for space heating in transmission [g/s]
p_0	Dead state pressure [bar]
$\dot{Q}_h$	Hot water demand [W]
$\dot{Q}_h$	Space heat preheating demand [W]
$\dot{Q}_s$	Space heating demand [W]
$\dot{Q}_{thp}$	Heat pump heating transmission demand [W]
$\dot{Q}_{th}$	Hot water transmission demand [W]
$\dot{Q}_{tp}$	Space heat preheating transmission demand [W]
$\dot{Q}_{ts}$	Space heating transmission demand [W]
$\dot{S}_{\mathsf{gen}}$	Entropy generation rate [W/K]
s	Specific entropy [kJ/kg K]

s_0 Dead state specific entropy [kJ/kg K]

ΔT_{df} Distribution forward temperature difference [K]

ΔT_{dr} Distribution return temperature difference [K]

T_0 Dead state temperature [K]

t_c Heat pump condenser temperature [°C]

t_e Heat pump evaporator temperature [°C]

t_{cd} Heat pump compressor discharge temperature [°C]

t_{df} Distribution forward temperature [°C]

t_{dr} Distribution return temperature [°C]

t_{hf} Hot water forward temperature [°C]

t_{hh} Hot water temperature after district heating [°C]

t_{hi} Fresh water temperature [°C]

t_{hp} Temperature after preheating [°C]

t_{ht} Hot water tank temperature [°C]

t_{sf} Space heating forward temperature [°C]

t_{sr} Space heating return temperature [°C]

t_{tf} Transmission forward temperature [°C]

t_{tr} Transmission return temperature [°C]

$\dot{W}$ Electric power [W]

x_{co} Heat pump condenser outlet vapor quality [–]

x_{ei} Heat pump evaporator inlet vapor quality [–]

x_{eo} Heat pump evaporator outlet vapor quality [–]

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